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A COMPUTATIONAL STUDY OF EFFICIENCY OF CONTROLLABLE SEMIACTIVE MAGNETORHEOLOGICAL DAMPERS REDUCING LATERAL VIBRATION OF ROTORS WORKING IN CHAMBERS SUBMERGED IN LIQUIDS

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Key words: Rotor, Submerged Chamber, Semiactive Magnetorheological Squeeze Film Damper, Additional Masses and Damping, Coupled Physical Problems, Computational Methods.

Abstract. The design of rotating machines working in chambers submerged in liquids is a complicated technological problem. Vibration of the whole system produced by unbalance of the rotating parts can be significantly reduced if damping devices are inserted between the rotor and its stationary part. Due to variable operating conditions depending on the depth of the chamber submerging to achieve their optimum performance, the damping effect must be controllable. This is enabled by application of semiactive magnetorheological squeeze film dampers. Their damping is produced by squeezing a thin film of magnetorheological fluid between two rings and is controlled by the change of the magnetic flux passing through the lubricating layer. The liquid, in which the chamber is submerged, is considered as incompressible and acts on the chamber walls by its inertia and damping effects. The investigating behaviour of such systems requires to analyse several mutually coupled physical problems (mechanical, hydraulical, electrical and magnetic).

1 INTRODUCTION

The design of rotating machines working in chambers submerged in liquids belongs to complicated technological problems. Unbalance of the rotors always produces their lateral vibration and forces transmitted through the coupling elements into the rotor housing and into the chamber walls. Their magnitude depends not only on the speed of the rotor rotation but also on the depth of submerging the chamber, the flow of the surrounding liquid and on position of the chamber in the tank or in the channel. The presence of the liquid shifts due to additional mass and damping the resonance frequencies and therefore also the critical revolutions of the rotor. Vibration of the whole system can be significantly reduced if damping devices are inserted between the rotor and its stationary part. Due to variable operating conditions, to achieve their optimum performance, their damping effect must be controllable. This is enabled by application of semiactive magnetorheological squeeze film dampers. Their damping is produced by squeezing a thin film of magnetorheological fluid filling the gap between the inner and outer damper rings. As resistance against the flow of magnetorheological liquid depends on magnetic induction, the damping effect can be controlled by the change of the current in the coils generating the magnetic field.

The regular experimental and theoretical research of squeeze film magnetorheological dampers started in the end of the 20th century. In the mathematical models the magnetorheological fluids are usually represented by Bingham or Bulkley-Herschel materials. Vibration and control of a flexible rotor attenuated by a magnetorheological damping device was studied by Wang et al. [1]. The mathematical models of a long squeeze film magnetorheological damper based on modification of the Reynolds equation and on the assumption that the rotor journal exhibits a circular orbit can be found in [2], [3] and [4]. These works were extended by Zapomel and Ferfecki [5], [6] who developed the mathematical models of short and long magnetorheological dampers applicable also in the cases when trajectory of the rotor journal centre in the damper is a curve of a general shape and consequently applied them for computational simulations investigating the transient response of a rigid rotor passing the critical speeds [7].

A computational procedure for analysis of lateral vibration of a rotor submerged in perfect liquid is reported by Zapomel in [8], [9]. The pressure distribution in the liquid is described by a Laplace equation that is obtained by simplification of the Navier-Stokes equations and the equation of continuity. Components of the resulting force acting on the disc of the rotor are obtained by integration of the pressure distribution around the circumference and along the width of the disc. These forces can be expressed as a linear combination of the components of the disc acceleration and therefore these coefficients of proportionality are considered as additional masses. To solve the Laplace equation a finite element method was adopted.

The procedure presented in this paper is intended for investigating the transient response of rigid rotors working in chambers submerged in liquids focusing on attenuation of the rotor vibration. Its application requires to solve several mutually coupled physical problems (mechanical, hydraulical, electrical, magnetic).

2 THE INVESTIGATED SYSTEM

The investigated rotor RT works in a chamber CH placed in a tank TA filled with liquid (water) LQ (Fig.1). The rotor consisting of a shaft and of one disc is supported by rolling element bearings whose outer races are coupled with the bearing housings by spring elements B1, B2. The chamber is coupled with the wall of the tank by spring elements C1 and C2 exhibiting also some damping. The rotor turns at constant angular speed and is loaded by its weight and by the disc unbalance. The springs supporting the rotor are prestressed to eliminate their deflection caused by the rotor weight. The chamber is cylindrical and is partly or fully submerged in the liquid. It is loaded by its weight and by the forces transmitted from the rotor through the bearings and the damping elements into the walls of the chamber faces.

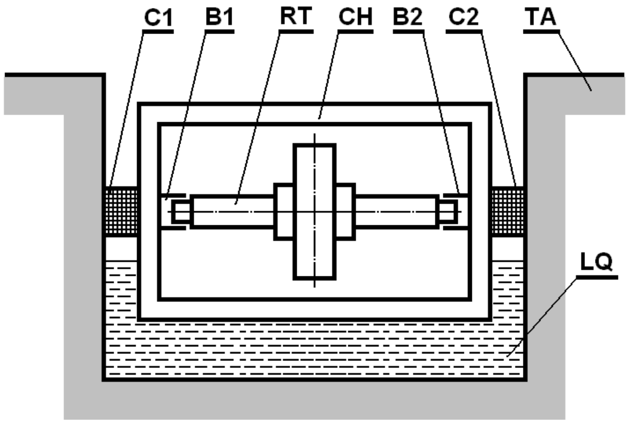

Figure 1: The investigated system

The task is to analyze influence of the change of damping of the magnetorheological dampers inserted between the rotor and its stationary part fixed with the chamber on attenuation of the system vibration at different operating conditions.

In the mathematical model the rotor, the chamber and the tank are considered as absolutely rigid bodies. The magnetorheological dampers placed between the rotor and its stationary part are represented by force couplings and the liquid in the tank is modelled as inviscid. Because of the design arrangement, the system is considered as symmetric relative to the middle plane of the disc and therefore the flow in the tank is two dimensional.

3 DETERMINATION OF THE DAMPING FORCE

The squeeze film magnetorheological damper consists of two rings between which there is a thin film of magnetorheological liquid (Fig.2). Both rings are coupled with the stationary part of the rotating machine, the outer ring directly, the inner one by a spring. The shaft is supported by a rolling element bearing whose outer race is coupled with the inner ring of the damper. Vibration of the inner ring relative to the outer one squeezes the film of magnetorheological liquid, which produces the damping force. In the body of the damper there is embedded an electric coil generating magnetic filed. The magnetic flux passes through the layer of magnetorheological liquid and as resistance against its flow depends on magnetic induction, magnitude of the current can be used to control the damping effect.

The development of the mathematical model of the damping element is based on assumptions of the classical theory of lubrication with the exception of modelling the lubricating liquid. The fixed and movable rings are assumed to be absolutely rigid and their surfaces absolutely smooth, the width of the gap between the rings is very small relative to the rings radii, the magnetorheological liquid is represented by Bingham material, its yield shear stress depends on magnetic induction, the flow in the lubricating layers is laminar and isothermal and the pressure in the radial direction remains constant and curvature of the oil film does not influence the lubricant flow.

The further attention will be focused only on the dampers whose geometry and design make possible to consider them as short [10]. Then the flow induced in the circumferential direction can be neglected.

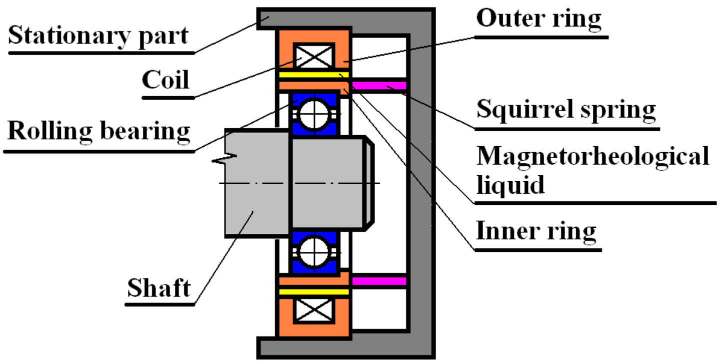

Figure 2: Scheme of the magnetorheological damper

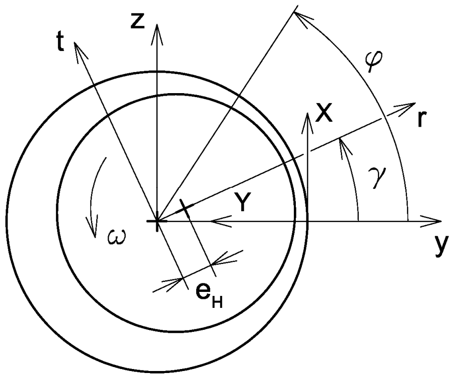

Figure 3: Coordinate system of the damper

The thickness of the lubricating film depends on position of the rotor journal centre relative to the system stationary part [10]

$$h = c - e_H \cos(\varphi - \gamma)\,. \tag{1}$$

h denotes the thickness of the oil film, c represents the width of the gap between the fixed and movable rings, e_H is the rotor journal eccentricity, φ is the circumferential coordinate and γ denotes the position angle of the line of centres (Fig.3).

To describe the pressure field in the layer of magnetorheological fluid, the Reynolds equation was modified for the case of Bingham liquid [6]

$$h^3 p'^3 + 3\left(h^2 \tau_y - 4\eta_B \dot{h} Z\right) p'^2 - 4\tau_y^3 = 0 \qquad \text{for} \qquad p' < 0\,, \tag{2}$$

$$h^3 p'^3 - 3\left(h^2 \tau_y + 4\eta_B \dot{h} Z\right) p'^2 + 4\tau_y^3 = 0 \qquad \text{for} \qquad p' > 0\,. \tag{3}$$

Equations (2) and (3) represent cubic algebraic equations with respect to the pressure gradient in the axial direction p'. η_B denotes the Bingham dynamical viscosity, τ_y is the yield shear stress, Z is the axial coordinate and ($\dot{}$) denotes the first derivative with respect to time.

As follows from the derivation, the root that has the physical meaning must satisfy the following conditions:

- it must be real (not complex),
- the conditions of validity of equations (2) and (3) must be satisfied, this means that the real roots obtained from (2) must be negative and the real ones obtained from (3) must be positive,

- $p' < -\dfrac{2\tau_y}{h}$ for $p' < 0$, (4)

  $p' > \dfrac{2\tau_y}{h}$ for $p' > 0$. (5)

The pressure profile p is given by integration of the pressure gradient

$$p = \int p' \, dZ . \tag{6}$$

Equations (2) and (3) are solved with the boundary condition expressing that pressure at the damper faces is equal to the pressure in the ambient space

$$p = p_A \qquad \text{for} \qquad Z = \pm\frac{L}{2} . \tag{7}$$

L is the length of the damper.

It is assumed that in the areas where the thickness of the lubricating film rises with time ($\dot{h} > 0$) a cavitation occurs. Pressure of the medium in cavitated regions remains constant and equal to the pressure in the ambient space. Then it holds for the pressure distribution in the film of the magnetorheological oil

$$p_D = p \qquad \text{for} \qquad p \geq p_A \qquad \text{and} \qquad p_D = p_A \qquad \text{for} \qquad p < p_A . \tag{8}$$

In the simplest design case of the damper, the rings, between which there is a layer of magnetorheological liquid, can be considered as a divided core of an electromagnet. Then the dependence of the yield shear stress on magnetic induction can be approximately expressed

$$\tau_y = k_y \left(\frac{N\,I}{2h} \right)^{n_y} . \tag{9}$$

k_y and n_y are the material constants of the magnetorheological liquid, N is the number the coil turns and I is the electric current.

y and z components of the damping force F_{MRy}, F_{MRz} are then given by integration of the pressure distribution around the circumference and along the length of the damping element. As it is assumed its symmetry relative to the plane perpendicular to the shaft centre line the resulting relations take the form

$$F_{MRy} = -2R \int_0^{2\pi} \int_0^{\frac{L}{2}} p_D \cos\varphi \, dZ \, d\varphi , \quad F_{MRz} = -2R \int_0^{2\pi} \int_0^{\frac{L}{2}} p_D \sin\varphi \, dZ \, d\varphi . \tag{10}$$

R is the radius of the gap between the damper rings.

4 THE FORCE OF THE LIQUID ACTING ON THE CHAMBER

To determine the force by which the liquid acts on the chamber it is assumed that the walls of the tank and of the chamber are absolutely rigid and smooth, the liquid is incompressible, newtonian, the flow is isothermal and due to the system symmetry it is two dimensional (2D).

Then the pressure and the velocity field is governed by the Navier-Stokes equations and by the equation of continuity

$$\frac{\partial p_F}{\partial x_1} + \rho_F \left(\frac{\partial v_1}{\partial t} + v_1 \frac{\partial v_1}{\partial x_1} + v_2 \frac{\partial v_1}{\partial x_2} \right) - \eta_F \left(\frac{\partial^2 v_1}{\partial x_1^2} + \frac{\partial^2 v_1}{\partial x_2^2} \right) = \rho_F g \,, \tag{11}$$

$$\frac{\partial p_F}{\partial x_2} + \rho_F \left(\frac{\partial v_2}{\partial t} + v_1 \frac{\partial v_2}{\partial x_1} + v_2 \frac{\partial v_2}{\partial x_2} \right) - \eta_F \left(\frac{\partial^2 v_2}{\partial x_1^2} + \frac{\partial^2 v_2}{\partial x_2^2} \right) = \rho_F g \,, \tag{12}$$

$$\frac{\partial v_1}{\partial x_1} + \frac{\partial v_2}{\partial x_2} = 0 \,. \tag{13}$$

x_1, x_2 are the y and z cartesian coordinates, v_1, v_2 are the velocity components of the flow in the y and z directions, p_F is the pressure, ρ_F, η_F are the density and dynamical viscosity of the liquid, g is the gravity acceleration and t is the time.

As the chamber performs oscillations only with small amplitude, the Reynolds number of the flow is low and the liquid inertia forces are dominant [11], [12]. On these conditions the viscous and convective terms in the Navier Stokes equations can be neglected and consequently (11) - (13) can be transformed into a Laplace equation describing the pressure distribution in the liquid [12]

$$\frac{\partial^2 p_F}{\partial x_1^2} + \frac{\partial^2 p_F}{\partial x_2^2} = 0 \,. \tag{14}$$

Because of the performed simplifications, p_F in (14) covers only the pressure induced by vibration of the chamber but it does not include the hydrostatic component that is caused by the weight of the fluid and that produces the lifting force acting on the chamber.

The liquid particles touching the body surface are not constrained in the tangential direction and in the normal direction they move together with the chamber wall. This is expressed by the boundary condition needed for solving the Laplace equation (14)

$$\frac{\partial p_F}{\partial n} = -\rho_F \, a_n \,. \tag{15}$$

n is the coordinate in the direction the outer normal to the chamber surface and a_n is acceleration of the liquid particle on the boundary in the direction of the outer normal.

As the chamber exhibits only a sliding motion, the boundary condition (15) can be rewritten in the following form

$$\frac{\partial p_F}{\partial x_1} \cos \alpha_n + \frac{\partial p_F}{\partial x_2} \sin \alpha_n = -\rho_F \left(a_y \cos \alpha_n + a_z \sin \alpha_n \right). \tag{16}$$

a_y, a_z are the acceleration components of the chamber sliding motion in the y and z directions and α_n is the directional angle of the outer normal of the chamber surface.

The governing equation (14) together with the boundary condition (16) is solved by application of a finite element method. After discretization of the region filled with the liquid and after some manipulation described in details e.g. in [9], [12] one obtains at each point of time a set of linear algebraic equations for unknown values of the pressure at all nodes.

It follows from the Laplace equation (14) and the boundary condition (16) that the pressure can be expressed at each moment of time as a sum of terms proportional to the acceleration components of the sliding motion of the chamber

$$p_F(x_1, x_2) = a_y\ p_y^*(x_1, x_2) + a_z\ p_z^*(x_1, x_2) + p_0 . \tag{17}$$

p_y^*, p_z^* are the coefficients of proportionality that depend only on the border (the chamber wall) geometry and p_0 is the pressure component depending on the pressure at the free surface (atmospheric pressure). Then integration of the pressure distribution around the circumference and along the length of the submerged part of the chamber gives components of the hydraulic force acting on the chamber

$$F_{Fy} = -L_{CH} \int_{\Gamma_{CH}} p_F\ \cos\alpha_n ds\ , \qquad F_{Fz} = -L_{CH} \int_{\Gamma_{CH}} p_F\ \sin\alpha_n ds\, . \tag{18}$$

L_{CH} is the length of the chamber and Γ_{CH} is the border of the region filled with the liquid corresponding to the chamber wall.

Taking into account relation (17), equations (18) can be rewritten into the form [9], [12]

$$F_{Fy} = -m_{Fyy}\ a_y - m_{Fyz}\ a_z + F_{Fy0}\, , \tag{19}$$

$$F_{Fz} = -m_{Fzy}\ a_y - m_{Fzz}\ a_z + F_{Fz0}\, . \tag{20}$$

Coefficients m_{Fyy}, m_{Fyz}, m_{Fzy}, m_{Fzz} are called additional masses and the force components F_{Fy0}, F_{Fz0} depend on the pressure on the free surface. The additional masses express inertia effects of the liquid by which the vibration of the chamber is influenced. They depend on the mutual position of the chamber relative to the tank and on geometric shape of their walls.

5 THE EQUATIONS OF MOTION OF THE INVESTIGATED SYSTEM

Lateral vibration of the rotor and of the chamber partly or fully submerged in water is described by four equations of motion. Taking into account the system symmetry, they have the form

$$m_R \ddot{y}_R + 2k_{CHR} y_R - 2k_{CHR} y_{CH} = m_R e_T \omega^2 \cos(\omega t + \psi_0) - 2F_{MRy}\, , \tag{21}$$

$$m_R \ddot{z}_R + 2k_{CHR} z_R - 2k_{CHR} z_{CH} = m_R e_T \omega^2 \sin(\omega t + \psi_0) - 2F_{MRz} - m_R g + 2F_{PS}\, , \tag{22}$$

$$m_{CH} \ddot{y}_{CH} + 2b_{CHT} \dot{y}_{CH} - 2k_{CHR} y_R + 2(k_{CHR} + k_{CHT}) y_{CH} = 2F_{MRy} + F_{Fy}\, , \tag{23}$$

$$m_{CH} \ddot{z}_{CH} + 2b_{CHT} \dot{z}_{CH} - 2k_{CHR} z_R + 2(k_{CHR} + k_{CHT}) z_{CH} = 2F_{MRz} + F_{Fz} - 2F_{PS} - m_{CH} g + F_L \tag{24}$$

where

$$2F_{PS} - m_R g = 0\, . \tag{25}$$

m_R, m_{CH} denote the mass of the rotor and of the chamber, k_{CHR}, k_{CHT} are stiffnesses of the coupling elements placed between the rotor and the chamber and the chamber and the tank, b_{CHT} is the coefficient of linear damping in the constraint between the chamber and the tank,

F_{MRy}, F_{MRz} are components of the damping force produced by the magnetorheological dampers in y and z directions, F_{Fy}, F_{Fz} are the y and z components of the hydraulic force by which the liquid acts on the chamber, F_{PS} is the prestress force, F_L is the lifting force, e_T is eccentricity of the rotor centre of gravity, ω is angular speed of the rotor rotation, ψ_o is the phase lag, y_R, z_R, y_{CH}, z_{CH} are displacements of the rotor and chamber centres in the y and z directions respectively and $(\dot{})$, $(\ddot{})$ denote the first and second derivatives with respect to time.

After some manipulations the equations of motion can be rewritten into a matrix form

$$\mathbf{M}\,\ddot{\mathbf{x}} + \mathbf{B}\,\dot{\mathbf{x}} + \mathbf{K}\,\mathbf{x} = \mathbf{f}_{CO} \cos \omega t + \mathbf{f}_{SO} \sin \omega t + \mathbf{f}_{MR} + \mathbf{f}_F + \mathbf{f}_G + \mathbf{f}_L \,. \tag{26}$$

M, **B**, **K** are the mass, damping and stiffness matrices, $\mathbf{f}_{MR}$ is the vector of damping forces produced by the magnetorheological dampers, $\mathbf{f}_F$ is the vector of hydraulic force by which the liquid acts on the chamber, $\mathbf{f}_{CO}$, $\mathbf{f}_{SO}$ are the vectors of the unbalance force acting on the rotor, $\mathbf{f}_G$, $\mathbf{f}_L$ are the vectors of gravitational and lifting forces and **x** is the vector of displacements of the rotor and of chamber centres.

To solve the equation of motion (26) a modified Newmark method has been applied. To simplify the calculation procedure, the damping forces related to time t+Δt are determined by means of their expansion into a Taylor series in the neighbourhood of time t and by neglecting the terms of the second and higher orders

$$\mathbf{f}_{MR,t+\Delta t} = \mathbf{f}_{MR,t} + \mathbf{D}_B\left(\dot{\mathbf{x}}_{t+\Delta t} - \dot{\mathbf{x}}_t\right) + \mathbf{D}_K\left(\mathbf{x}_{t+\Delta t} - \mathbf{x}_t\right) + \ldots \;. \tag{27}$$

$\mathbf{D}_B$ and $\mathbf{D}_K$ denote the Jacobi matrices of partial derivatives.

After these manipulations the equation of motion (26) referred to time t+Δt is rewritten

$$\begin{aligned} &\mathbf{M}\,\ddot{\mathbf{x}}_{t+\Delta t} + \left(\mathbf{B} - \mathbf{D}_{B,t}\right)\dot{\mathbf{x}}_{t+\Delta t} + \left(\mathbf{K} - \mathbf{D}_{K,t}\right)\mathbf{x}_{t+\Delta t} = \\ &= \mathbf{f}_{CO} \cos \omega t + \mathbf{f}_{SO} \sin \omega t + \mathbf{f}_{MR,t} - \mathbf{D}_B \dot{\mathbf{x}}_t - \mathbf{D}_K \mathbf{x}_t + \mathbf{f}_F\left(\ddot{\mathbf{x}}_{t+\Delta t}\right) + \mathbf{f}_G + \mathbf{f}_L \,. \end{aligned} \tag{28}$$

According to (18) - (20) components of the hydraulic force F_{Fy}, F_{Fz} can be expressed by means of the chamber acceleration components

$$\mathbf{f}_{F,t+\Delta t} = -\mathbf{M}_F\,\ddot{\mathbf{x}}_{t+\Delta t} + \mathbf{f}_{P0} \,. \tag{29}$$

Consequently the equation of motion (26) related to time t+Δt takes the form

$$\begin{aligned} &\left(\mathbf{M} + \mathbf{M}_F\right)\ddot{\mathbf{x}}_{t+\Delta t} + \left(\mathbf{B} - \mathbf{D}_{B,t}\right)\dot{\mathbf{x}}_{t+\Delta t} + \left(\mathbf{K} - \mathbf{D}_{K,t}\right)\mathbf{x}_{t+\Delta t} = \\ &= \mathbf{f}_{CO} \cos \omega t + \mathbf{f}_{SO} \sin \omega t + \mathbf{f}_{MR,t} - \mathbf{D}_B \dot{\mathbf{x}}_t - \mathbf{D}_K \mathbf{x}_t + \mathbf{f}_G + \mathbf{f}_L + \mathbf{f}_{P0} \,. \end{aligned} \tag{30}$$

$\mathbf{M}_F$ is the matrix of additional mass and $\mathbf{f}_{P0}$ is the vector of the hydraulic force depending on the pressure at the free surface. As the displacements are considered to be small, the change of the shape of the region filled with the water is negligible and coefficients of matrix $\mathbf{M}_F$ remain constant. The resulting equation of motion (30) has the form required by the basic Newmark algorithm and its calculation arrives at each integration step at solving a set of linear algebraic equations. But the effective stiffness matrix must be repeatedly set up.

To perform solution of the equation of motion (30) it is assumed that at the beginning the system is in rest and takes the equilibrium position.

6 COMPUTATIONAL SIMULATIONS

The task was to investigate the influence of the damping effect of the magnetorheological dampers on the vibration amplitude of the rotor and of the chamber in the tank with water for two specified speeds of the rotor rotation (50 rad/s and 70 rad/s) and for the cases when the chamber is not submerged and is submerged to 1/3 of its radius.

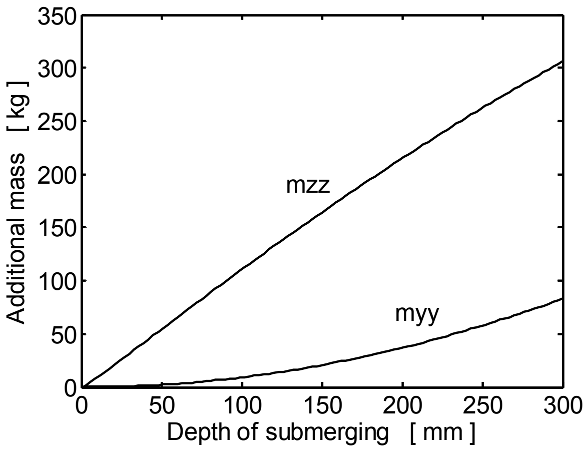

Figure 4: Additional masses

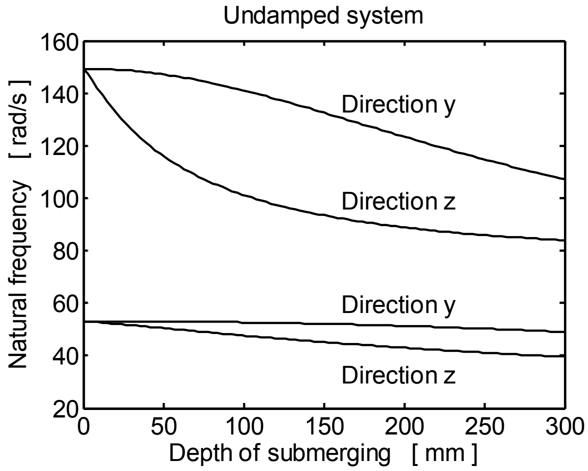

Figure 5: The submerged system natural frequencies

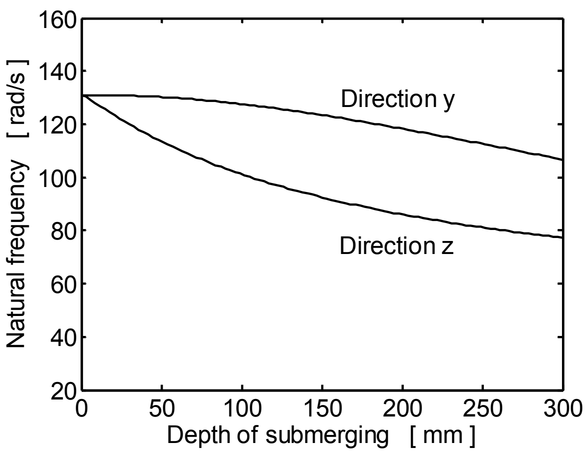

Figure 6: The overdamped system natural frequencies

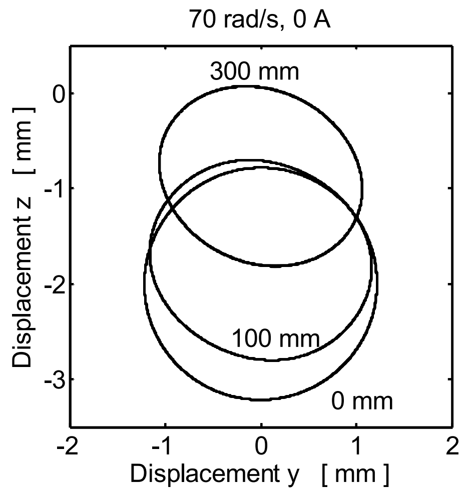

Figure 7: Positions of the chamber centre orbits

In Fig.4 one can see that the additional masses related to the horizontal (m_{yy}) and vertical (m_{zz}) directions grow up with increasing depth of submerging. Rising magnitudes of the additional masses arrives at the decrease of the system undamped natural frequencies (Fig.5). The splitting of the courses is caused by different magnitudes of the additional masses in the horizontal and vertical directions. If the coupling between the rotor and the chamber is strongly overdamped, then the rotor and the chamber behave as one rigid body. The dependence of its natural frequencies on the submerging depth is drawn in Fig.6. As evident the change of damping in the magnetorheological dampers changes the system modal properties and shifts its natural frequencies. This can be utilized for attenuation of the oscillation amplitude.

In Fig.7 one can see the affect of the lifting force on the orbits of not submerged (0 mm) and partly submerged (100 mm, 300 mm) chamber. The trajectory centers are pushed upward to the free water surface.

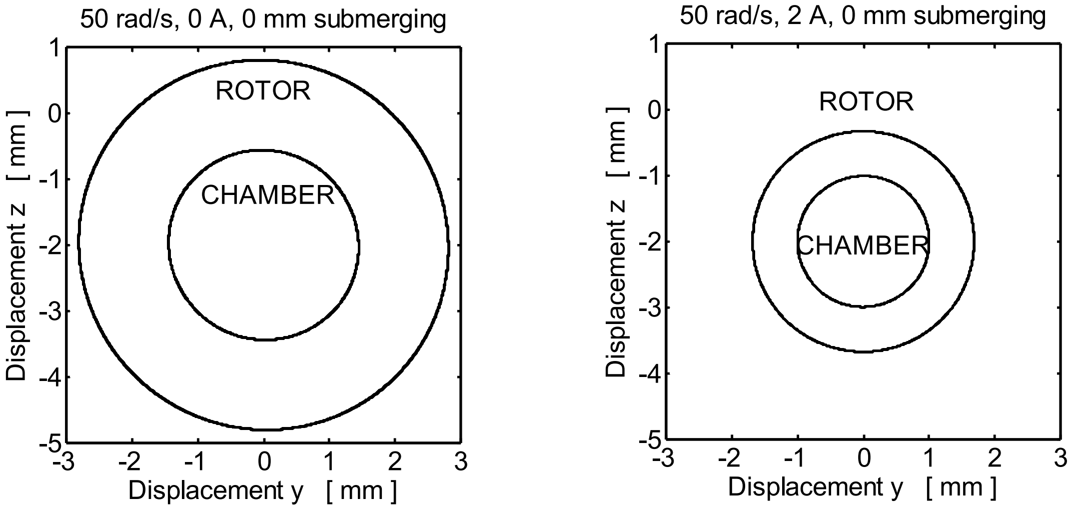

Figure 8: Orbits of not submerged chamber at speed of the rotor rotation 50 rad/s

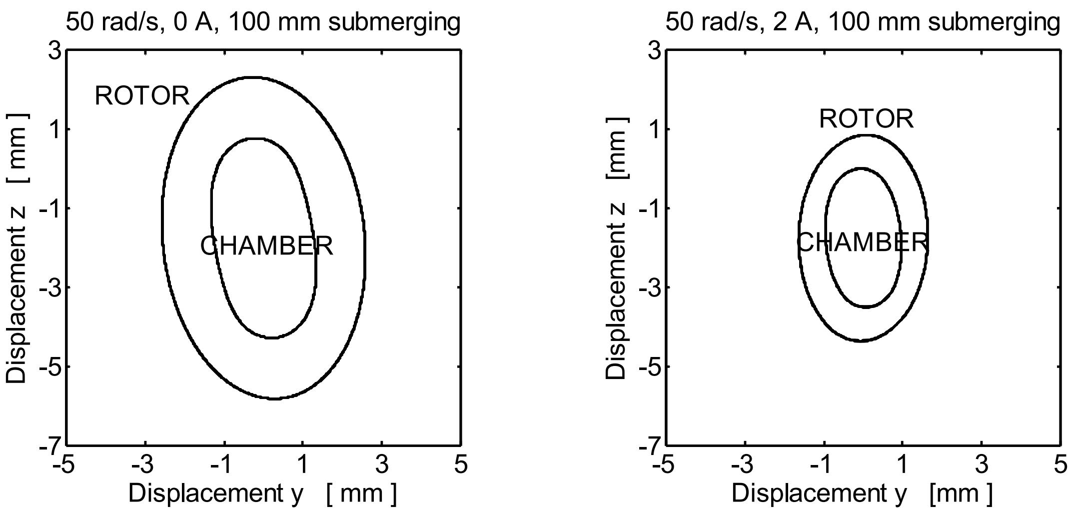

Figure 9: Orbits of submerged chamber at speed of the rotor rotation 50 rad/s

In Fig. 8 to 10 there are drawn orbits of the rotor and chamber centres for increasing magnitudes of the current controlling the damping effect of the magnetorheological dampers. Its evident that for the speed of the rotor rotation 50 rad/s rising damping (rising magnitude of the applied current) arrives at higher reduction of the oscillation amplitude. Comparing results in Fig. 8 and 9, the displacements are larger if the chamber is partly submerged. Submerging the chamber also changes the shape of the orbits. Fig. 10 shows a different behaviour of the system. The rising current leads to increase of the size of orbits of the rotor and the chamber centres.

The observed phenomena can be explained by the change of two system parameters : amount of damping in the magnetorheological dampers and influence of the inertia effects of the liquid (additional mass of the water), in which the chamber is submerged, and by nonlinear character of the magnetorheological damping force. These parameters lead to changing the course of the resonance curves and to shifting the resonance peaks. The obtained results confirm that to reach the minimum amplitude of the system vibration the damping effect must be controllable to be possible to be adapted to the current operating conditions.

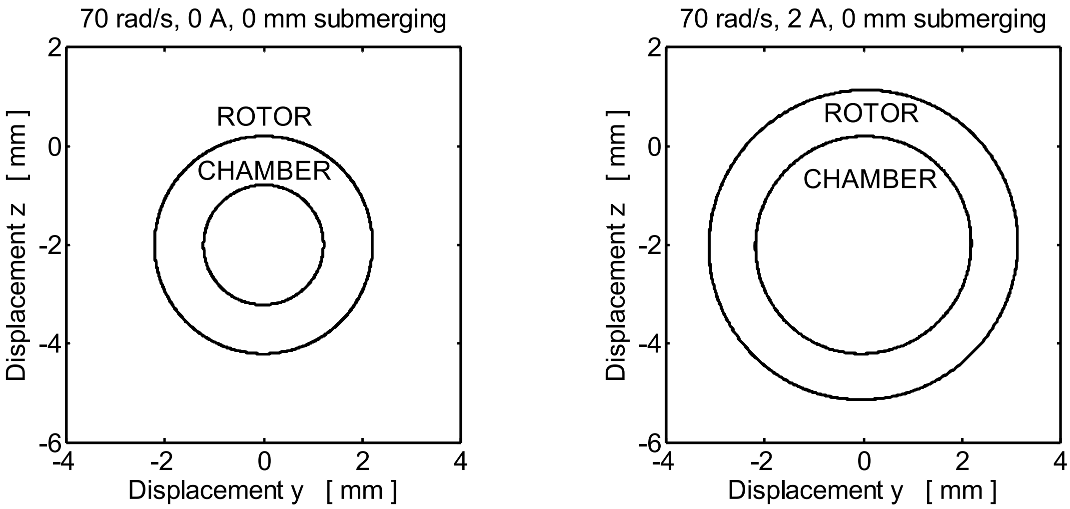

Figure 10: Orbits of not submerged chamber at speed of the rotor rotation 70 rad/s

7 CONCLUSIONS

The developed procedure represents a tool for analysis of the transient and steady state vibration of rigid rotors placed in chambers submerged in liquids. The variable depth of submerging changes the system parameters, which has the influence on the system resonance frequencies and critical speeds of the rotor rotation. Amplitude of the induced vibration can be significantly reduced if damping devices are inserted between the rotor and its stationary part fixed with the chamber. To achieve their optimum performance, the damping effect must be controllable. For this purpose application of squeeze film magnetorheological dampers were investigated. As the damping effect depends on magnitude of the magnetic flux passing through the layer of magnetorheological fluid, the change of the current generating the magnetic field can be used for its control. Results of the performed computational simulations confirm that the magnetorheological dampers make possible to adapt the damping effect to the current operating conditions and to minimize amplitude of the system vibration. Advantage of the proposed damping mechanism is that it always produces some amount of damping, which if needed, can be increased. In addition it does not require a complicated and expensive control system and makes possible to achieve the optimum performance of the rotating machine by means of adapting the damping effect to the current operating conditions.

Investigating attenuation of vibration of rotating machines working in chambers submerged in liquids by means of application of magnetorheological dampers is a complicated technological problem. Its solving requires to analyze several mutually coupled physical problems (mechanical, hydraulical, electrical, magnetic).

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REFERENCES

[1] Wang, J. and Meng, G. and Hahn, E.-J. Experimental study on vibration properties and control of squeeze mode MR fluid damper-flexible rotor system. *Proceedings of the ninth Engineering Conference,* Chicago, (2003).

[2] Forte, P. and Paterno, M. and Rustighi, E. A magnetorheological fluid damper for rotor applications. *International Journal of Rotating Machinery* (2004) **10**:175-182.

[3] Forte, P. and Paterno, M. and Rustighi, E. A magnetorheological fluid damper for rotor applications. *Proceedings of the IFToMM Sixth International Conference on Rotor Dynamics*, Sydney, (2002).

[4] Wang, G.-J. and Feng, N. and Meng, G. and Hahn, E.-J. Vibration control of a rotor by squeeze film damper with magnetorheological fluid. *Journal of Intelligent Material Systems and Structures* (2006) **17**:353-357.

[5] Zapoměl, J. and Ferfecki, P. Mathematical modelling of a long squeeze film magnetorheological damper for rotor systems. *Modelling and Optimization of Physical Systems* (2010) **9**:97-102.

[6] Zapoměl, J. and Ferfecki, P. Mathematical modelling of a short magnetorheological damper. *Transactions of the VŠB – Technical University of Ostrava, Mechanical Series* (2009) **LV**:289-294.

[7] Zapoměl, J. and Ferfecki, P. A computational investigation of vibration attenuation of a rigid rotor turning at a variable speed by means of short magnetorheological dampers. *Applied and Computational Mechanics* (2009) **3**:411-422.

[8] Zapoměl, J. The approach to computational analysis of vibration of vertical rotors supported by fluid film bearings and having a disc submerged in an inwettable liquid. *Proceedings of the 8th World Congress on Computational Mechanics,* Venice, (2008), CD 1-2.

[9] Zapoměl, J. The procedure for numerical investigation of vibration of a rotor with fluid film bearings and a disc submerged in inwettable liquid. *Proceedings of the 9th International Conference on Vibrations in Rotating Machinery*, Exeter, (2008).

[10] Krämer, E. *Dynamics of rotors and foundations.* Springer-Verlag, (1993).

[11] Bathe, K.-J. *Finite Element Procedures in Engineering Analysis.* Prentice-Hall, (1982).

[12] Levy, S. and Wilkinson, J.P.D. *The component element method in dynamics with application to earthquake and vehicle engineering*. McGraw-Hill, (1976).